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Published in:
Energy

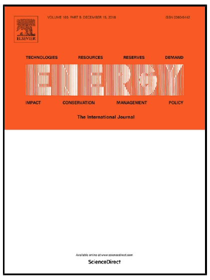
DOI (link to publication from Publisher):
10.1016/j.energy.2018.12.209

Publication date:
2019

Document Version
Accepted author manuscript, peer reviewed version

Citation for published version (APA):
Arabkoohsar, A. (2019). Non-uniform temperature district heating system with decentralized heat pumps and standalone storage tanks. *Energy*, *170*, 931-941. https://doi.org/10.1016/j.energy.2018.12.209

Accepted Manuscript



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PII: S0360-5442(18)32583-0

DOI: 10.1016/j.energy.2018.12.209

Reference: EGY 14462

To appear in: *Energy*

Received Date: 08 September 2018

Accepted Date: 27 December 2018

Please cite this article as: A. Arabkoohsar, Non-uniform temperature district heating system with decentralized heat pumps and standalone storage tanks, *Energy* (2018), doi: 10.1016/j.energy.2018.12.209

Non-uniform temperature district heating system with decentralized heat pumps and standalone storage tanks

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Abstract

In this work, the novel concept of non-uniform temperature district heating (NUTDH) system with decentralized heat pumps and standalone heat storage units (HPHS) is proposed. In the NUTDH-HPHS system, the temperature within the transmission pipeline is always at the ultra-low level of 35-40 ^{o}C, which is sufficient for space heating use. The heat pumps will increase the temperature within the distribution pipes to 70 ^{o}C during a short period of time a day. This temperature is to provide the domestic hot water (DHW) need of the consumers. Heat pumps are sized in neighborhood scale, and as each neighborhood is supplied with a high temperature just for a short time a day, each heat pump may be assigned for a few neighborhoods. As not always high-temperature water is available, the substations are equipped with storage tanks. In this system, the rate of heat loss is minimal, legionella risk is absolutely zero, and there is a strong synergy between the power and heat sectors. The system is designed and analyzed for a case study. The results are also compared with the performance of other popular district heating (DH) schemes. It is demonstrated that the NUTDH-HPHS system shows the best performance.

Keywords: District heating; Non-uniform supply temperature; Decentralized heat pump, standalone storage unit; Stratified storage tank.

1. Introduction

DH is a key element of smart energy systems (SES) [1]. According to [2], in an SES, there must be concrete synergies between the DH system and the other energy sectors, especially electricity. As such, DH is a high capacity sector for utilization of renewable sources, where a further certain feature of the SES is the high integration rate of renewable energy [3]. Today, lower-temperature DH (LTDH) systems are emerging to facilitate the utilization of low-grade renewable technologies, such as geothermal and evacuated tube solar systems, etc., and increasing the heat production efficiency procedure [4].

There is a quite extensive literature on the feasibility of low-temperature and even ultralow- temperature (ULTDH) systems. The two main objectives of decreasing the supply temperature in a DH system is decreasing the rate of loss and increasing the potential for utilization of more renewable sources [5]. There are, however, a number of challenges on the way of developing an LTDH or a ULTDH system. The risk of legionella generation and the inability for DHW preparation are two of the main challenges of such systems. Yang et al. [6] presented a 3E (energy, exergy and economic) analysis of various solutions for DHW supply with LTDH and ULTDH systems. Olsen et al. [7] presented a sort of propositions for facilitating the development of a 50 °C-LTDH system. Wang et al. [8] proposed and analyzed the combination of high- and low-temperature DH systems for energy-efficient buildings. The use of heat pumps in the substations was another solution offered for DHW preparation in a ULTDH system [9]. Ommen et al. [10] investigated several different configurations of substations with a heat pump. Ostergaard et al. investigated the use of booster and central heat pumps in LTDH systems for DHW preparation [11]. Nord et al. [12] studied the challenges and potentials for LTDH system implementation in Norway and found the linear heat density and return water temperature playing crucial roles in the transition to LTDH system. Yang and Svendsen [13] modeled and analyzed different substations for a ULTDH system, and offered a number of solutions for enhancing the energy efficiency of heat delivery process. In another work, Yang and Svendsen [14] carried out a thorough analysis and feasibility study of using booster and central heat pumps in ULTDH systems with low heat density and analyzed the DH system from energy, exergy and economy performance points of view. Im and Liu [15] investigated the feasibility of a bi-lateral heat trade model in LTDH systems, proved the sufficiency of their trade model and provided a reliable reference for energy network design and planning. Rama and Sipila [16] studied the effects of the transition to the LTDH in existing DH systems, the needed technical solutions and the improved potential of utilizing appropriate heat sources. Kauko et al. [17] studied the dynamic performance of the LTDH systems with the dynamic simulation program Dymola, and provided a thorough report of their results. Lund and Mohammadi [18] studied the required characteristics of the insulation materials for an LTDH system. Gadd and Werner studied the challenge of achieving low return temperature from the substation of LTDH systems and proposed a novel method for this based on a fast detection of differential temperature faults [19].

An LTDH system with high integration of renewable sources, with strong synergy with other energy sectors, and with two-way heat exchange with the end-users is so-called a 4th generation of DH (4GDH) system [20]. Lund et al. [21] have thoroughly outlined the main characteristics of a 4GDH. Kamal [22] discussed the practical methods of integrating the existing DH systems with the 4GDH systems. Ziemele et al. [23] analyzed the feasibility of a 4GDH system in Latvia and assessed various policies. Paiho and Reda [24] did the same for Finland and discussed the challenges on the way of transition for the current DH (3GDH) system to the 4GDH system. Tereshchenko and Nord [25] studied the possible components of the heat production chain of a 4GDH systems considering various technologies, the economic aspects, and the technical limitations. The essential improvements required on the design and construction strategies of a 4GDH systems are brought up in [26]. Volkova et al. [27] analyzed the barriers of transition to the 4GDH systems and presented a methodology for this transition process. Sameti and Haghighat [28] presented an optimal design of a 4GDH system focusing on energy reciprocity and investigated the effects of heat and electricity exchange on the cost-effectiveness and emission level of such a system.

In this work, the new scheme of NUTDH with decentralized heat pumps and standalone heat storage tanks (NUTDH-HPHS) is proposed and thermodynamically analyzed. In this system, the transmission pipeline of the DH network is always at the low temperature of 35-40 ^{o}C. Decentralized neighborhood-scale heat pumps increase the temperature of the distribution network of each neighborhood to 70 ^{o}C for a short time every day. The heat pumps are sized based on the demand of one neighborhood, and each heat pump could supply a few neighborhoods in turn. Because of the lack of hot enough water when the heat pumps are not in operation for a given neighborhood, the substations are equipped with standalone heat storage units. The duration of the operation of the heat pumps for high-temperature supply for each neighborhood is in relation to the size of the storage tanks which should be defined based on professional optimization techniques. In this work, however, the storage tanks are sized based on the Danish standards and the heat pumps come into operation whenever the heat storage units run out of hot water. There is a detailed explanation of this system in the next section. In order to prove the proficiency of the DH scheme proposed, it is designed and analyzed for a case study in Denmark.

It bears mentioning that the main novelty of this work is the special design of the NUTDH-HPHS system that makes it unique yet highly efficient and compatible with all the features of the 4GDH systems. In addition, while the main gap of the DH knowledge is the technical drawbacks and limitations of the LTDH and the ULTDH designs that have restricted the implementation of these schemes, the NUTDH-HPHS concept does not suffer from any of these limitations and may be considered as a highly feasible solution for filling this gap. For proving this claim, the results of the simulations on the NUTDH-HPHS system are compared with those obtained for other DH schemes, i.e. ULTDH, LTDH, and 3GDH systems, in the same case study. It will be shown how the NUTDH-HPHS system performs better than all the other DH options in various aspects.

2. The case study

This work is conducted based on the energy and comfort regulations of Denmark. Therefore, a case study in this country, for which the required database of the project is available, is appropriate for that. The required database includes the DHW and space heating demand profiles, comfort standards etc. for the network. As it is very difficult to have access to the demand profile of the consumers of the DH networks due to the security reasons, the alternative method is to calculate the demands, for which the characteristics of the buildings, solar energy availability, ambient temperature information, etc. are required. A small network in the city of Aarhus, for which this sort of data is available, has been taken as the case study of this work.

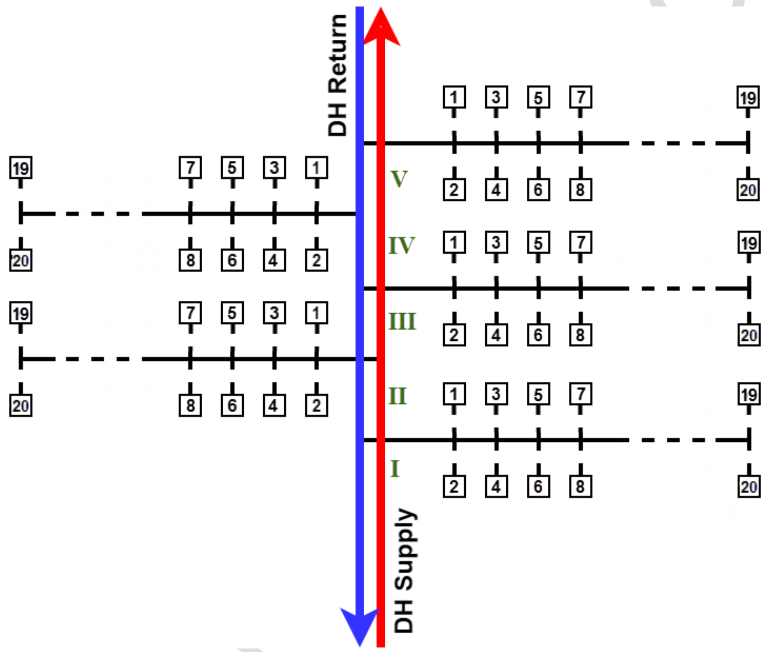

Fig. 1 The arrangement of the end-users in the DH network.

In this case study, a total of 100 single-family detached houses are considered to be covered by the DH system. Figure 1 shows the topology considered for the buildings and the pipeline of the case study DH system. As seen, the network comprises of 5 streets with 20 buildings in each. Here, the length of the transmission pipeline (I) is assumed to be 1500 m, the length of pipes between the streets (II, III, IV and V) is 100 m, the distance of the houses in each street is 20 m. The living area of each building is 150 m^2 on a rectangular field of 15 m × 10 m.

The space heating demand of the buildings is calculated based on the characteristics considered for them as well as the environmental conditions. According to [29], the Danish comfort indoor temperature is 20 ^{o}C, which is considered for the indoor temperature of the buildings within the network. According to [30], an average overall heat loss factor of 0.93 W/m².K is considered for the buildings. This is based on 2.32 W/m².K for windows (20% of the building shell surface), 0.625 W/m².K for walls (30%), 0.5

W/m².K for roofs (25%) and 0.625 W/m².K for the floor including cold bridges (25%). As all the buildings in the network are considered to be residential, the average hourly air exchange rate for ventilation is 0.65 m³/h per m³ of the heated building volume. It is also noteworthy that the average daily internal gain of the building due to the inhabitants' metabolism effects is considered as 2.3 ^{o}C [30].

The Danish Standard DS-439 [31] states that DHW supply temperature should be at least 45°C. For making a DHW consumption pattern for the network, the standard Danish draw-off profile, given in Table 1, is used. This draw-off profile is, however, randomized for all of the buildings to make a reasonable simultaneity factor of Draw off over the entire network. In order to make this demand profile even more similar to a real case, the randomization was so managed that the distribution of the draw-offs is at least three times larger during the two 6-hour periods of 6:00-11:59 o'clock and 18:00-23:59 o'clock. Note that tapping temperature of 40 ^{o}C is seen in the table which means the hot water supplied by the system (at least 45 ^{o}C) is mixed with domestic cold water to provide the required temperature for eahc specific draw-off.

Table 1 Draw-off profile of a typical Danish single-family house/apartment without bathtub [31].

Time	Volume (lit)	Temperature (^{o}C)	Duration (min)
10:55	42	40	5
10:55	15	45	2,5
11:05	0	0	0
11:15	42	40	5
11:15	15	45	2,5
11:25	0	0	0
11:35	42	40	5
11:45	0	0	0
11:55	42	40	5
///////	///////	///////	///////
22:55	42	40	5
22:55	15	45	2,5
23:05	0	0	0
23:15	42	40	5
23:15	15	45	2,5
23:25	0	0	0
23:35	42	40	5
23:45	0	0	0
23:55	42	40	5

The given profile out of this randomization, along with the space heating demand profile of the network, will be presented in the results section.

3. The NUTDH vs. other popular DH schemes

In this section, the NUTDH- HPHS, along with the other DH schemes that are going to be compared to each other, is introduced and detailed information about each of these schemes is presented.

3.1. The 3GDH system

The 3GDH system is, which is the current DH system of many locations in the world, based on a supply temperature of 70-85 °C and a return line temperature of 35-45 °C [32]. There are two substation configurations in this DH scheme, namely the instantaneous heat exchange unit (IHEU) and the district heating storage unit (DHSU). In an IHEU, there are two plate heat exchangers for DHW and space heating supply, while in the DHSU; one buffer tank is added to the DHW supply section for peak shaving. This storage tank causes a significant reduction in the size of the pipeline and the DH system equipment. In Denmark, the standard capacity of IHEU substation is 32 kW, whereas this value for a DHSU substation is only 4 kW. This big difference is just due to the sharp DHW draw-off that happens few times a day only and the storage tank offers a shaving service for that [7]. Note that, regardless of the substation type, if the pressure of the DH pipeline is below 10 bar, the heat exchanger of the space heating section may be removed so that the supplied DH water may go directly through the in-building heat distribution systems. The sketch diagram of these substation types, with marked standard Danish temperature values in the primary and secondary sides, is presented in Figure 2 (the IHEU in Figure 2-a, and the DHSU in Figure 2-b). In this figure, T_A is the temperature of the normal water within the water piping of the building, so-called domestic cold water (DCW). T_A is considered to be equal to the indoor temperature.

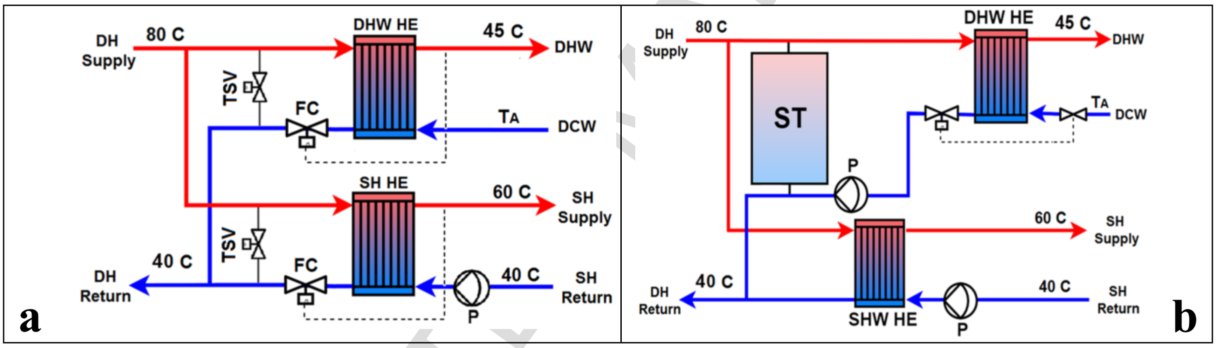

Fig. 2 Sketch diagram of an IHEU (a) and a DHSU (b) substations; HE: heat exchanger, TSV: thermostatic valve, FC: flow controller, P: pump, SH: space heating.

3.2. The LTDH system

In an LTDH system, the supply and return temperatures are 50-55 °C and 25-30 °C, respectively [7]. The rest of the system characteristics, e.g. the piping system, heat suppliers, etc., are just similar to the 3GDH system. This scheme is much appropriate for Denmark, as this supply temperature range is enough for DHW preparation in this country, while this cannot be enough for the Netherlands, for instance, in which the minimum required temperature of DHW is 60 °C. Due to this fact, much research and applied projects have been conducted on this scheme in Denmark. The LTDH demonstration projects reported in [7] are only a few of the investigations carried out. For this DH scheme, due to the low supply temperature, there is no need for any storage unit. Thus, the only possible substation configuration for this scheme is an IHEU, just similar to Figure 2-a, with the DH supply and return temperatures of 55 and 25 °C.

3.3. ULTDH system

Again, with the same main characteristics as the previously introduced DH schemes, in a ULTDH system, the supply and the return temperature ranges are 35-45 °C of 25-30 °C. Although this temperature range is appropriate for efficient preparation of space heating demand, it is not sufficient for direct DHW supply. Therefore, either this system is for space heating purpose only or a specific substation design is required for that for providing the DHW need of the consumers. In the former case, the DHW demand is provided by the individual electrical heater units (IEHU substation), and in the latter case, employing standalone heat pumps is one of the most popular proposals, so-called as heat pump furnished units (HPFU). Figure 3 illustrates these two substations, i.e. an IEHU (3-a), and an HPFU (2-b). As seen in Figure 3-b, the hot water supplied by the DH is used for space heating and the DHW demand is provided by an electrical coil within a buffer tank. In Figure 3-b, it is seen that a heat pump does DHW preparation task increasing the DH supply temperature to the comfort temperature. A thorough explanation of various HPF substation designs may be found in [10].

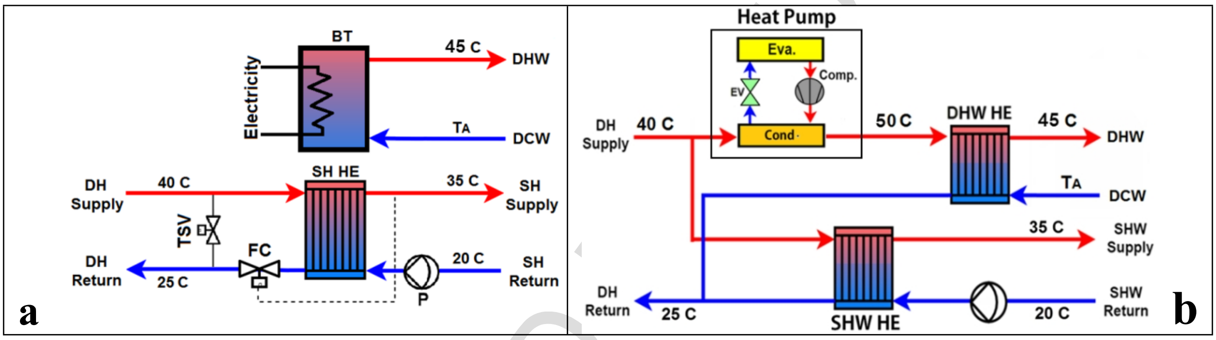

Fig. 3 Sketch of the ULTDH system substations; the IEHU (a) and the HPFU substations (b), BT: buffer tank.

3.4. The NUTDH-HPHS system

As explained before, the proposed NUTDH-HPHS system works based on an ultra-low temperature in transmission pipeline all the time, and through the distribution pipes over a major time of the day. The decentralized neighborhood-scale heat pumps will increase the temperature of supply for each neighborhood from the ultra-low level to a moderate level during a few hours a day for each neighborhood. The substations in this scheme are equipped with a storage tank to store heat for DHW use for the time that heat pump is not supporting the neighborhood. Therefore, having this system, the DHW need of the network can be sufficiently provided, the network offers a very low-rate of loss, the energy conversion efficiency will be high and there will be a smart integration between the heat and the electricity sectors. In addition, the heat pumps offer thermal disinfection for the entire piping in the system so that the legionella risk is totally prevented. Based on the operation strategies designed for this DH scheme:

- The space heating demand is supplied by the ultra-low temperature level of 40 °C, while the medium-temperature of around 70 °C is applied for a short period during the day for DHW preparation.

- The upper and lower design temperatures are chosen based on the comfort standard temperatures, the temperature drop along the pipeline, and the expected efficiency of the heat exchangers.
- Decentralized heat pumps, in neighborhood scale and each heat pump for supporting a few neighborhoods, are employed for increasing the ultra-low supply temperature to the desired medium temperature. Figure 4 shows the configuration of the a DH network based on a NUTDH design, in which one heat pump is allocated for supporting the five neighborhoods in turn. Note that here the figure shows various types of buildings while the case study is based on similar buildings. The figure, indeed, shows the possibility of employing such a system for any kind of end-user.

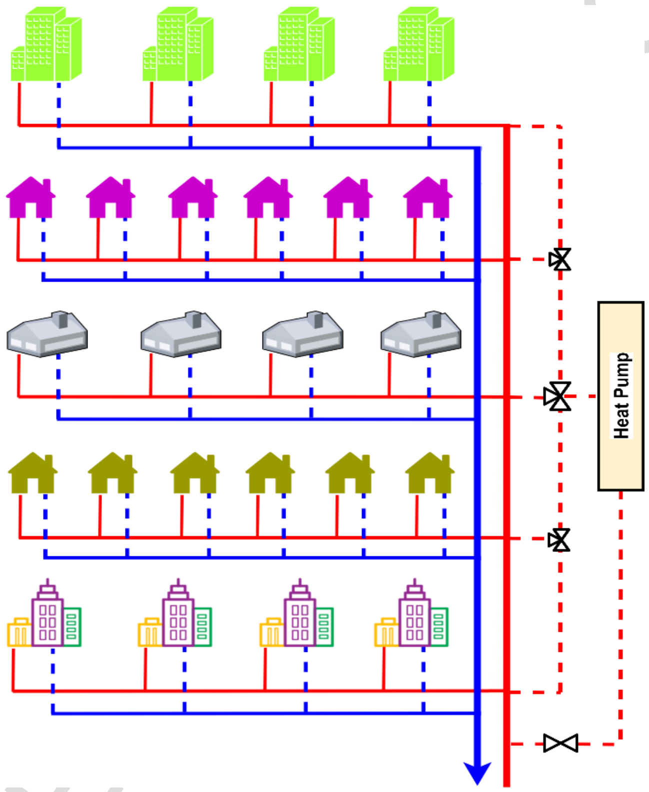

Fig. 4 The schematic of an NUTDH-HPHS system; red: supply line, blue: return line, dashed-lines: flows from/into the heat pump.

- As medium-temperature supply is only available a few hours a day, heat storage tanks are required at the substations. The substation of the consumers should be equipped with storage tank units. In this work, helically coiled heat storage units are employed. Overall, the substation comes with the configuration shown in Figure 5.

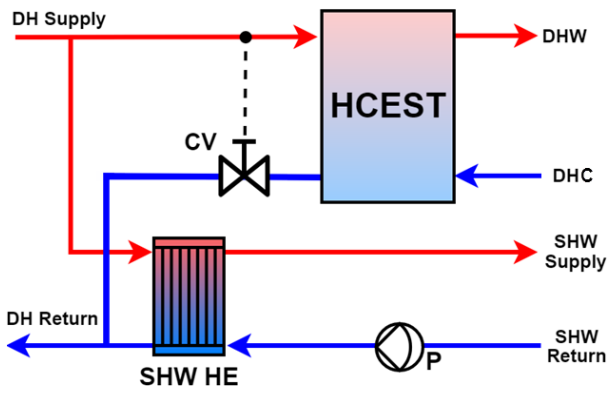

Fig. 5 The schematic of a substation in an NUTDH-HPHS system; HCEST: helically coiled energy storage tank.

- Figure 6-left shows the schematic of the storage tank. The dimensions and characteristics considered for the tank and for the helix are those reported in [33]. Figure 6-right illustrates the method that the DH flow through the spiral coil is controlled during the charging mode. According to the figure on the right panel, the control valve operates as a function of the inlet and outlet flows of the DH side of the storage tank so that the bigger the temperature difference is the more the valve opens and vice versa.

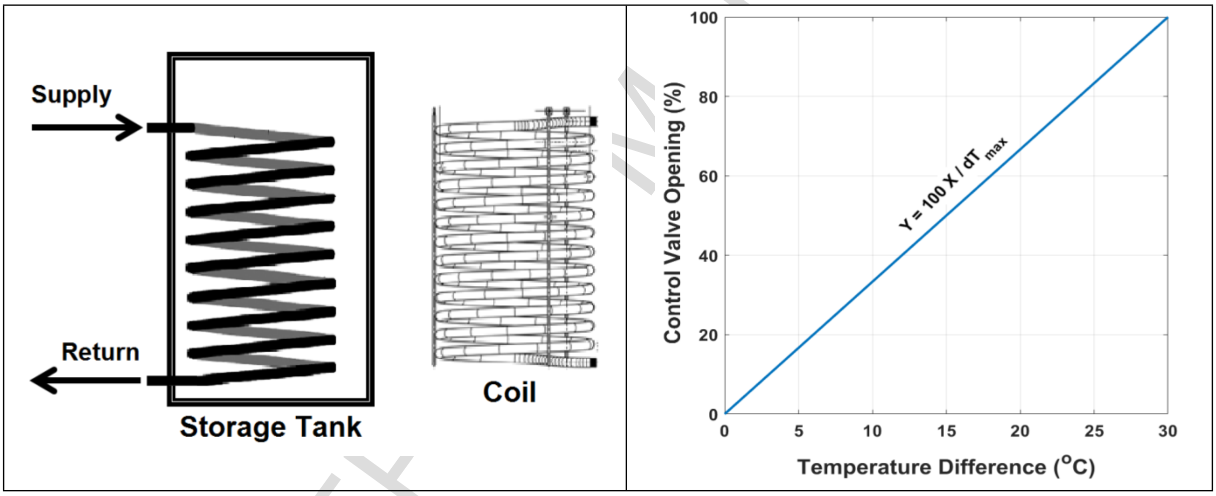

Fig. 6 The storage tank configuration (left), and the method of the flow control through the helix (right).

Based on the defined control strategy, the DH water charges all the tanks until they are fully charged. As the charging process continues and the temperature difference between the top and the bottom of the tank gets lower, the valve gradually closes. Immediately after having a uniform temperature along the storage tank, the control valve closes and the charging process ends.

3.5. Summary of the various DH schemes characteristics

In the end, besides the main characteristics and standards of the in-building heating systems, Table 2 presents information about the characteristics considered for various DH schemes discussed above. As seen, some features are common between all or a few of cases and others are only for a specific case.

Table 2 The main characteristics of different DH schemes.

Parameter	Information			
	3GDH	LTDH	ULTDH	NUTDH
Design supply temperature (^oC)	80	55	40	70/40
Design return temperature (^oC)	40	25	25	30/25
Space heating temperatures (^oC, inlet/outlet)	70/40	50/25	35/25	60/40 or 35/25
Substation design (DHW/SH)	IHEU or DHSU/IHEU	IEHU /IHEU	IEHU or HPFU/IHEU	HCEST/IHEU
Nominal pressure (barg)	12			
Pipe type	Twin [34]			
Insulation class	Series II and series III [34]			
Maximum media speed (m/s)	2			
DHW supply temperature (^oC)	45			
Room comfort temperature (^oC)	20			

4. Thermodynamic Model

The thermodynamic model of a DH system, most of which is common for all the various discussed DH cases is presented hereunder.

The first step of modeling a DH system is calculating the rate of heat demand in the network and sizing the pipeline based on the maximum load of the system. The heat demand for DHW depends upon the hot water draw-off of the network buildings. In this work, a randomized vector of Danish standard single-family DHW draw-off profile was employed for the buildings. The energy demand for space heating, on the other hand, is a function of several parameters, such as the comfort temperature, the ambient temperature, the building stuck energy performance, etc. The following equation calculates the rate of demand for space heating in each building:

$$\begin{aligned}\dot{Q}_{sh,b} = & \overbrace{\rho_a V_{a,b}\left({T_{in}}^{\lambda+1} - {T_{in}}^{\lambda}\right)}^{\dot{Q}_{s,b}} + \overbrace{\rho_a \dot{V}_{ven}\left(T_{in} - T_{out}\right) + UA_{l,b}\left(T_{in} - T_{out}\right)}^{\dot{Q}_{l,b}} \\ & + \overbrace{\rho_{bm} V_{bm}\left({T_{bm}}^{\lambda+1} - {T_{bm}}^{\lambda}\right)}^{\dot{Q}_{std,b}} - \overbrace{\sum_{n=1}^{M}\left(A_n \mathrm{I_T} (\tau\alpha)_{avg}\right)}^{\dot{Q}_{g,b}}\end{aligned} \quad (1)$$

in which, T is temperature, V is volume, ρ represents density. The subscriptions *a*, *in*, *out*, *b* and *bm* refer, respectively, to the air within the building, the indoor condition, the ambient condition, the building and the building stuck material. The superscription λ is the time step counter. As marked, each of the statements on the right side of the equation represents an energy flow coming in or going out of the building. The first term on the right is the rate of heat required to increase the indoor temperature. The summation of the second and the third terms calculate the total rate of heat losses from the building due to the ventilation and heat loss to the ambient. In these two terms, $\dot{V}_{ven}$ is the volume flow rate of air replaced by fresh air for ventilation, and $UA_{l,b}$ is the overall heat loss coefficient of the building through the walls and windows. The fourth term indicates the rate of energy stored in the building stuck which is zero in steady state condition, and the last item calculates the rate of heat gain of the building due to the solar irradiation. In this item, I_T is the solar irradiation through the windows, A_n is the area of each of the windows and $(\tau\alpha)_{avg}$ refers to the average transmission-absorption coefficient of the windows and the internal elements of the building exposed to the sun rays. M is the total number of apertures letting sun rays coming into the building.

Next, calculating the rate of heat losses through the pipeline should be accomplished. The rate of heat loss from a piece of pipe, with x m length, can be calculated by:

$$\dot{Q}_l(x) = UA_l(x)(T_{in} - T_s) \tag{2}$$

in which, T_{in} is the temperature of the fluid within the pipe, T_s is the temperature of the surrounding. UA_l is the overall heat loss coefficient that can be given as:

$$UA_l(x) = \left(\frac{1}{h_{in}\pi d_{in} x} + \frac{ln\left(\frac{r_{out}}{r_{in}}\right)}{2k_p \pi x} + \frac{ln\left(\frac{r_{ins,out}}{r_{ins.in}}\right)}{2k_{ins}\pi x} \right)^{-1} \tag{3}$$

where, r_{out}, r_{in}, $r_{ins,out}$ and $r_{ins,in}$ refer to the external and internal radiuses of the pipe and the insulation, respectively. k_p and k_{ins} are the conductivity factors of the pipe and the insulation material. Finally, h_{in} is the convective heat transfer coefficient, which considering a turbulent flow through the pipe, may be obtained from [35]:

$$h_{in} = \frac{0.023 Re_D^{0.8} Pr^{0.4} k}{d_{in}} \tag{4}$$

in which, Re_D is Reynolds number, Pr is Prandtl number, k is the conductivity of the fluid and d_{in} is the internal diameter of the pipe.

Dividing the pipe into several small divisions with x m length each and constant temperature, assuming a constant surface temperature internal flow, the temperature of water through the next divisions is calculated as:

$$T_e = (T_i - T_s)\, exp\left(\frac{UA_l x}{\dot{m}c}\right) + T_s \tag{5}$$

where, T_e is the outlet temperature of the fluid in the previous pipe division (the inlet of the next division), T_i is the inlet temperature of the previous division. $\dot{m}$ is the mass flow rate, and c is the thermal capacity of water.

Calculating the heat demand of the network and the temperature drop along the path, one could size the pipeline of the district heating system. The pipeline should be such sized that it may carry the maximum heat load of the network based on the maximum allowed velocity through the pipes, i.e. 2 m/s. Then, the diameter of the pipe is calculated by:

$$d_{n,p} = \left(\frac{4\dot{Q}_{dh,max}}{\rho u_{max} c \Delta T}\right)^{0.5} \tag{6}$$

where, $d_{n,p}$ is the nominal diameter of each section of the pipeline, $\dot{Q}_{dh,max}$ is the maximum heat load of the given pipeline section, u_{max} is the maximum velocity of water in the pipe, and ΔT is the temperature difference between the return and supply lines.

For calculating the rate of pressure drop along the pipeline, two sources of pressure reduction should be considered. These are pressure losses because of friction (major) and the minor losses due to the other factors. It is recommended to consider the minor losses as 20% of the major losses in a water pipeline. Therefore, the total pressure loss in a DH pipeline with x m length could be calculated by:

$$\Delta P_t = \Delta P_{major} + \Delta P_{minor} = 1.2\ \Delta P_{major}; \textbf{ where: } \Delta P_{major} = \frac{fx\rho u^2}{2d_{in}} \tag{7}$$

in which, u is velocity and *f* is Darcy friction factor given by [35]:

$$f = \begin{cases} 0.316 Re_D^{-0.25} & if \quad Re_D \leq 20000 \\ 0.184 Re_D^{-0.2} & if \quad Re_D > 20000 \end{cases} \tag{8}$$

Having the value of the pressure losses, and marking the specific volume of water with $_v$, the work of the booster pumps in the network for compensating the pressure losses is calculated by [36]:

$$\dot{W}_p(x) = \dot{m} v \Delta P_t = \frac{0.6 f \dot{m} x u^2}{d_{in}} \tag{9}$$

The next component, which is an active element of a DH system at the substations, is the plate heat exchanger. Considering counter-flow plate heat exchangers, the following equations may be applied [37]:

$$\dot{Q}_{hx} = \varepsilon_{hx} \overbrace{C_{min}(T_{h,in} - T_{c,in})}^{\dot{Q}_{hx,max}} \tag{10}$$

$$\varepsilon_{hx} = \frac{1 - exp[-(UA_{hx}/C_{min})(1 - C_r)]}{1 - C_r exp[-(UA_{hx}/C_{min})(1 - C_r)]}; \boldsymbol{where}: C = \dot{m}c \textbf{ and } C_r = \frac{C_{min}}{C_{max}} \tag{11}$$

where, $\dot{Q}_{hx}$ is the heat transferred from the DH water to the secondary side of the heat exchanger. For the NUTDH-HPHS system, this item is equal to the space heating demand (Q_{sh}) of the building while it can also be equal to the DHW demand of the building if the substation has a further heat exchanger for DHW supply as well (this applies to other DH schemes). $\dot{Q}_{hx,max}$ is the maximum rate of heat that could be transferred between the fluids, $T_{h,in}$ is the inlet temperature of the hot side of the heat exchanger, i.e. DH supply line, and $T_{c,in}$ refers to the inlet temperature of the secondary fluid, i.e. the return line of the space heating or DHW supply loops. ε_{hx} is the effectiveness factor of the heat exchangers, UA_{hx} is the overall heat trasfer coefficient of the heat exchanger which is a function of the design and physical charactersitics of them, and the subscriptions *min* and *max* refer to the fluids with the lower and higher values of C, respectively. Clearly, having the inlet and outlet temperatures of the primary and secondary side fluids based on the standards and design objectives, one can simply calculate the mass flow rate of fluid in each side, and subsequently, the rate of heat transferred.

In addition to the common components employed in various DH schemes, there are two components that are specifically only for the NUTDH-HPHS system, i.e. the heat pump and the individual heat storage tanks. Regarding the heat pumps, the heating duty of the heat pump for each neighborhood ($\dot{Q}_{hp,n}$) could be calculated by:

$$\dot{Q}_{hp,n} = \dot{m}_{dh,n} c_p (T_{high} - T_{low}) = \dot{Q}_{l,n} \left(\frac{\beta_{hp}}{\beta_{hp} - 1} \right) \tag{12}$$

In which, $\dot{m}_{dh,n}$ is the mass flow rate of DH water required for the *n'th* neighborhood/street, T_{high} is the desired temperature after the heat pump (70 °C) and T_{low} is the temperature of DH water before the heat pump which is supposed to be at the ultra-low level. Also, β_{hp} is the heat pump coefficient of performance (considered as 4 here), and $\dot{Q}_{l,n}$ is the rate of heat that should be rejected from the heat pump working fluid to the low-temperature heat source of the heat pump, which can an air source or a ground source heat pump. The electricity required to derive the heat pump for producing this amount of heat is also calculated by:

$$E_{hp,n} = \dot{Q}_{hp,n} - \dot{Q}_{l,n} = \dot{Q}_{hp,n} \left(\frac{1}{\beta_{hp}} \right) \tag{13}$$

As mentioned, helically coiled heat storage tanks are going to be used for NUTDH-HPHS system. Figure 7 shows the schematic of this type of storage tank.

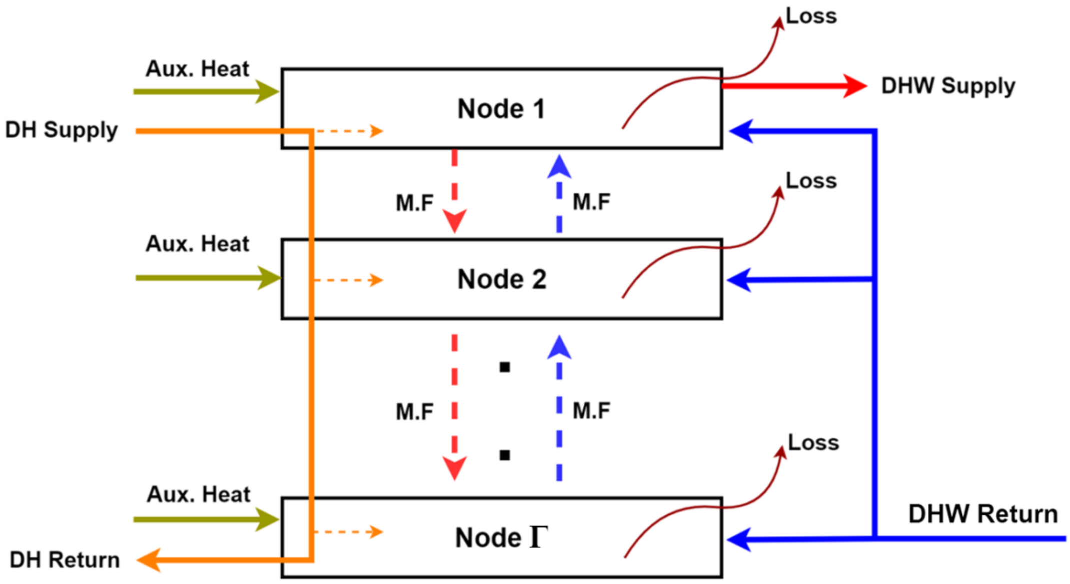

Fig. 7 The scheme of a stratified helically coiled heat storage tank; Aux. Heat: auxiliary heat, M.F: mixing flows.

Such a storage tank will operate with significant degrees of stratification so that the upper layers of the tank are hotter than the lower layers. For modeling a stratified tank, the tank is divided into Γ nodes and the energy balance equation is written for each of the nodes. The result will be a set of differential equations that should be solved to calculate the temperature of the nodes. The energy balance on the node j can be written as [33]:

$$\dot{Q}_{st,j}^{\lambda} = \dot{Q}_{sup,j}^{\lambda} + \dot{Q}_{ld,j}^{\lambda} + \dot{Q}_{mix,j}^{\lambda} + \dot{Q}_{l,j}^{\lambda} + \dot{Q}_{aux,j}^{\lambda} \tag{14}$$

in which, the superscription λ counts the time steps, and $\dot{Q}_{st,j}^{\lambda}$ is the rate of storage of heat into each node which may be explicitly calculable by:

$$\dot{Q}_{st,j}^{\lambda} = \rho c_p V_j \frac{T_j^{\lambda} - T_j^{\lambda-1}}{\Delta t} \tag{15}$$

where, V_j is the volume of water in each node, and Δt is the duration of each time step (1 min in this work). The rate of heat extracted from each node to provide the DHW demand is calculated by:

$$\dot{Q}_{ld,j}^{\lambda} = \dot{m}_{ld,j}^{\lambda} c_p \left(T_j^{\lambda} - T_{j+1}^{\lambda}\right) \tag{16}$$

where, $\dot{m}_{ld,j}^{\lambda}$ is the mass flow rate of the water extracted from the node j at each time step . The flow for providing the DHW demand comes from the first node and the same amount of water is replaced from the lower nodes to the upper ones. This mass flow rate is calculated by:

$$\dot{m}_{ld,j}^{\lambda} = \frac{\dot{Q}_{dhw}^{\lambda}}{c_p\left(T_{dhws}^{\lambda} - T_{dhwr}^{\lambda}\right)} \tag{17}$$

where, $\dot{Q}_{dhw}^{\lambda}$ refers to the heat extracted from the storage tank for DHW supply, T_{dhws}^{λ} is the temperature of the extracted water from the top node, and T_{dhwr}^{λ} is the temperature of the DHW return flow. The heat transfer from/into the *j'th* node due to the heat conductivity with the neighbor nodes is given by:

$$\dot{Q}^{\lambda}_{mix,j} = \frac{k}{\Delta y} A_{j+1}\left(T^{\lambda}_{j} - T^{\lambda}_{j+1}\right) + \frac{k}{\Delta y} A_{j-1}\left(T^{\lambda}_{j} - T^{\lambda}_{j-1}\right) \tag{18}$$

in which, Δy is the height of each node in the tank, k is the thermal conductivity of water, and A_j refers to the cross-section area of water in the tank.

$\dot{Q}^{\lambda}_{l,j}$ is the rate of heat losses from each node to the ambient during each time step and is calculated as:

$$\dot{Q}^{\lambda}_{l,j} = UA^{\lambda}_{l,j}\left(T^{\lambda}_{j} - T^{\lambda}_{a}\right) \tag{19}$$

$\dot{Q}^{\lambda}_{sup,j}$ is the heat supplied to each node through the helical coil. For calculating this parameter, the helical coil is divided into m nodes. For a better accuracy of the numerical model, the number of the coil elements should be larger than the opted storage tank node numbers. Thus, each tank node (j) houses a few coil elements (m). The energy balance of each of the coil nodes can be written as follow:

$$\rho c_p V_m \frac{\overline{T}^{\lambda}_{m} - \overline{T}^{\lambda-1}_{m}}{\Delta t} + \dot{m}^{\lambda}_{m} c_p\left(T^{\lambda}_{m+1} - T^{\lambda}_{m}\right) + UA^{\lambda}_{m}\left(\overline{T}^{\lambda}_{m} - T^{\lambda}_{m,ext}\right) = 0; \boldsymbol{where}: \overline{T}^{\lambda}_{m} = \frac{\left(T^{\lambda}_{m+1} + T\right.}{2} \tag{20}$$

in which, V_m is the volume of water within each node of the coil, $\dot{m}^{\lambda}_{m}$ is the mass flow rate of the DH water throug each node of the coil, T^{λ}_{m} is the temperature of water entering the coil node and T^{λ}_{m+1} is the water temperature leaving the coil node. Also, $\overline{T}^{\lambda}_{m}$ is the average temperature of water through the coil in the node m, and $T^{\lambda}_{m,ext}$ is the temperature of the external surface of the coil at node m. Therefore, the heat transferred from the coil to the node j of the tank, is calculated by:

$$\dot{Q}^{\lambda}_{sup,j} = \sum_{m=r}^{R} UA^{\lambda}_{m}\left(\overline{T}^{\lambda}_{m} - T^{\lambda}_{m,ext}\right); \boldsymbol{where}: T^{\lambda}_{m,ext} = T^{\lambda}_{j} \tag{21}$$

where, R-r is the number of coil nodes (m) lying in each node of the storage tank (j). Note that by applying the formulation given above for the stratified tank nodes, a set of algebraic nonlinear equations is obtained. Thus, an iterative solver based on tridiagonal matrix algorithm is used to solve the resulting equations system [38]. The helical coil is evaluated with a step-by-step model as presented in [39], considering a one-dimensional temperature distribution profile in the flow direction and using the control volume discretization technique.

5. Results and discussions

In this section, the results of the simulations and analysis carried out on the NUTDH-HPHS system for the Danish case study is presented.

Figure 8 presents information about the two environmental effective parameters on the performance of a DH system, i.e. the ambient temperature and the solar energy availability in the case study. As seen, the ambient temperature can be as low as 267 K sometimes during winter, which makes a high heat demand of the DH system, and a peak of about 303 K during summer. The solar irradiation is also high during summer with a peak of about 900 W on a horizontal surface while not much solar energy is expected during winter as Denmark is a far northern country with gloomy winters.

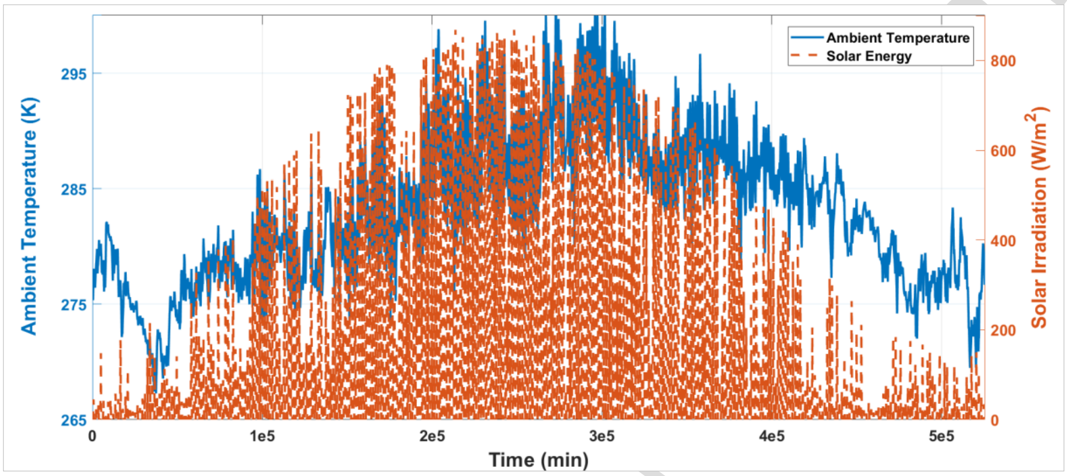

Fig. 8 Solar irradiation (right y-axis) and ambient temperature (left y-axis) in the case study over an entire year.

As discussed in section 2, a randomized version of the standard Danish hot water draw-off profile is considered for the buildings of the DH network in this work. As such, having the ambient temperature, the indoor comfort temperature, the solar energy availability, the buildings' characteristics, etc. one could make the calculation of the buildings heat demand for space heating. Figure 9, on the upper panel, shows the DHW draw-off pattern of a couple of randomly selected buildings in the network and the total energy demand of the DH system for DHW over a day. This daily pattern is repeated over the entire year. Figure 9, on the lower panel, shows the total heat demand of network for space heating along a whole year. As the buildings' characteristics and the other factors are considered the same for all the buildings their space heating demand is identical.

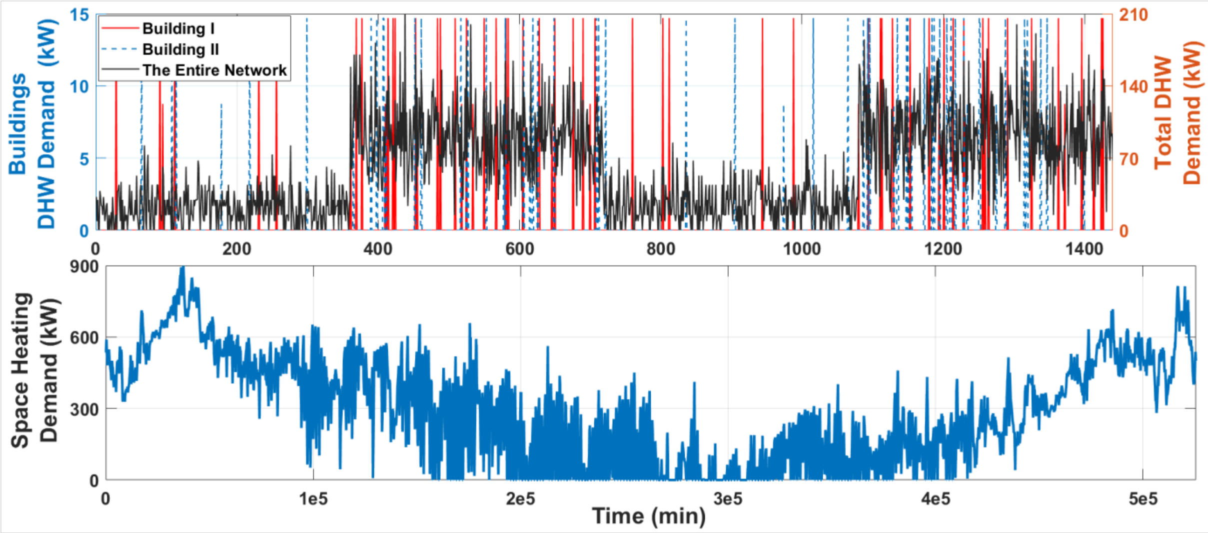

Fig. 9 Space heating and DHW demand profiles of the case study.

As seen, the network space heating demand reaches to the peak point of 900 kW during winter while this value fluctuates around very smaller values, with even absolute zero heat demand in many points, during the summer. The upper panel shows that how well the DHW profile is distributed so that it makes the two sample selected buildings to have their major draw-offs during the morning and before midnights but yet with low simultaneity. In this figure, the black fluctuating line gives the total network demand. As seen, the maximum of 210 kW is the DHW demand of the entire network which is sometimes a day reclaimed. Expectedly, much less DHW draw-off is observed during the night hours and the afternoon hours.

Having the randomized DHW draw-off profiles as well as the space heating demand profiles, one can calculate the rate of instantaneous total heat demand of the DH network and subsequently calculate the flow rate of water through the DH pipeline for each of the six considered scenarios. Figure 10 presents information about the total mass flow rate of the DH systems in the different cases for three consecutive winter and summer days with extreme low and high temperatures over the year. It is not surprising that the 3GDH system in both states results in the lowest flow rate as the difference between the supply and return temperatures is high there. On the other hand, the NUTDH-HPHS case gives the higher mass flow rate with a few peak periods every day for charging the storage tanks.

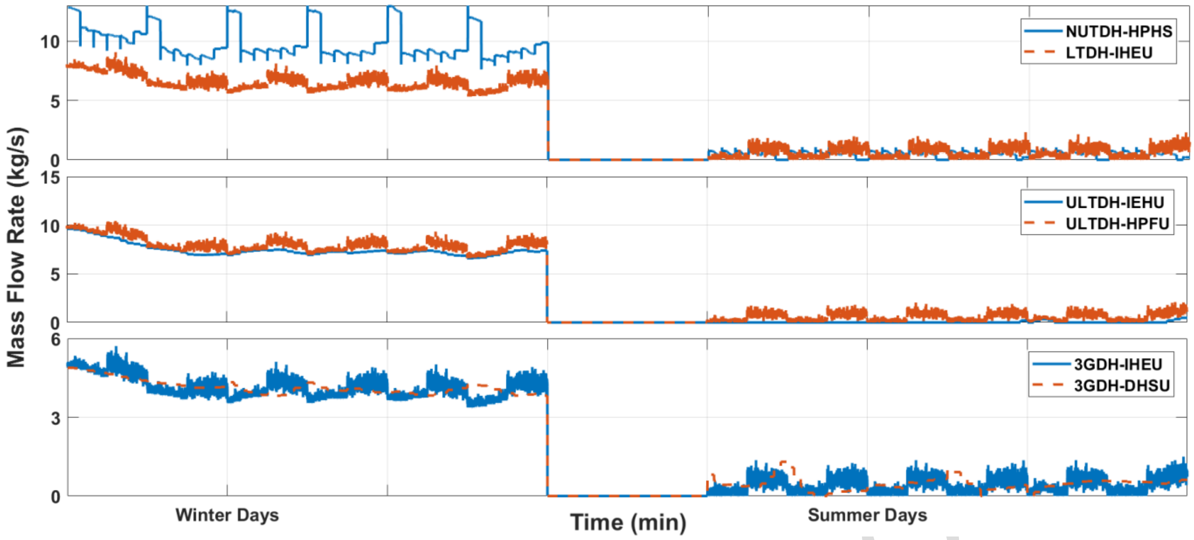

Fig. 10 The mass flow rate of the DH system in various scenarios.

Having the mass flow rate of the water required and the supply and return temperature in each scheme, one could size the transmission and distribution pipelines. Table 3 details the obtained diameters for each section of the pipe for each of the six schemes. Note that that for 3GDH schemes smaller house connection pipes are obtained, however, these values are adjusted to 10 mm as the smallest possible house connection pipes.

Table 3. The diameter of the pipes in different sections of various considered cases.

	3GDH		ULTDH		LTDH	NUTDH
	DHSU	IHEU	IEHU	HPFU	IHEU	HPHS
House Connection (Ø, mm)	10	10	10	13	11	13
Street Branch, 0-200 m (Ø, mm)	26	31	43	47	38	58
Street Branch, 201-400 m (Ø, mm)	18	24	30	35	29	41
Main Pipe, Part I (Ø, mm)	58	65	95	101	75	103
Main Pipe, Part II (Ø, mm)	52	58	85	91	67	94
Main Pipe, Part III (Ø, mm)	45	51	74	79	59	83
Main Pipe, Part IV (Ø, mm)	37	42	60	65	48	72
Main Pipe, Part V (Ø, mm)	26	33	43	49	38	43

As seen, a 3GDH system with a DHSU substation configuration results in the smallest pipe dimensions. The main reason for this is the peak shaving that the storage tanks provide throughout the system. On the other hand, the NUTDH-HPHS system needs the largest pipes in the network because of the sharp sudden increase in the heat load of the system when high-temperature supply starts and the storage tanks should rapidly charge. This can be modified through an optimization of the charging process in the heat storage units as well as the flow control valves operation strategies, which is the next part of this research project. The ULTDH system, in both substation configurations, also needs large pipes due to the small temperature difference of water in supply and return lines. The LTDH scheme needs pipe diameters smaller than the LTDH schemes and larger than the 3GDH systems.

Figure 11 illustrates the level of pressure drop along the entire pipeline of the considered cases during the typical winter and summer days. As seen, the rate of pressure loss in a ULTDH-IHEU system is far more than the others while NUTDH-HPHS and 3GDH-IHEU systems offer the lowest rate of pressure drops along the pipeline. Also, it is observed that the pressure loss rate is extremely higher during winter days where the water velocity within the pipes is evidently higher.

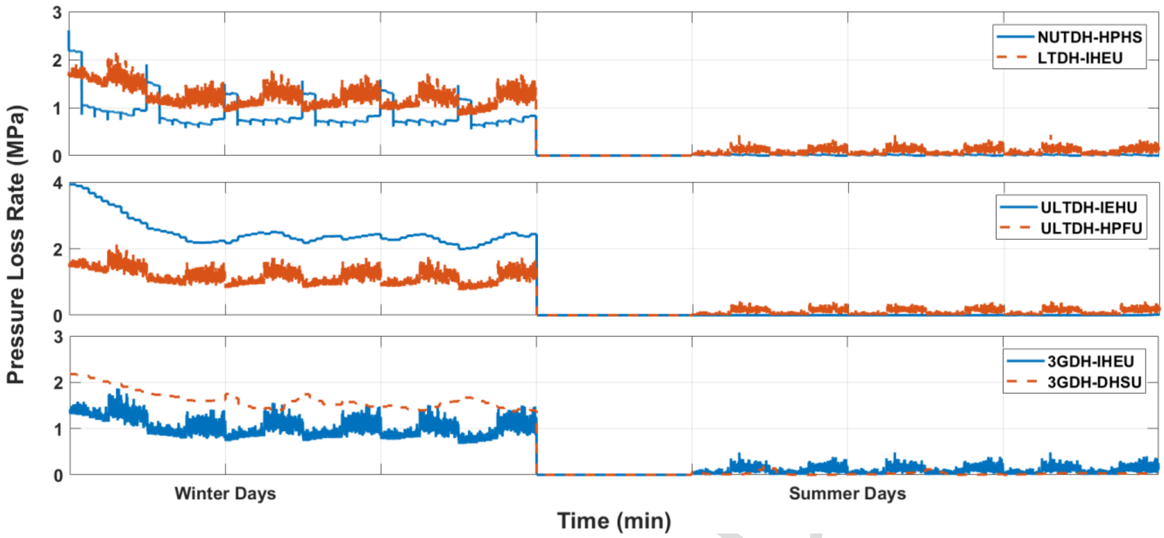

Fig. 11 The rate of pressure loss in different DH cases over the sample winter and summer days.

As, somehow, the most important factor in a DH system, Figure 12 presents information about the rate of heat loss in different DH schemes considered in this work over the typical winter and summer days.

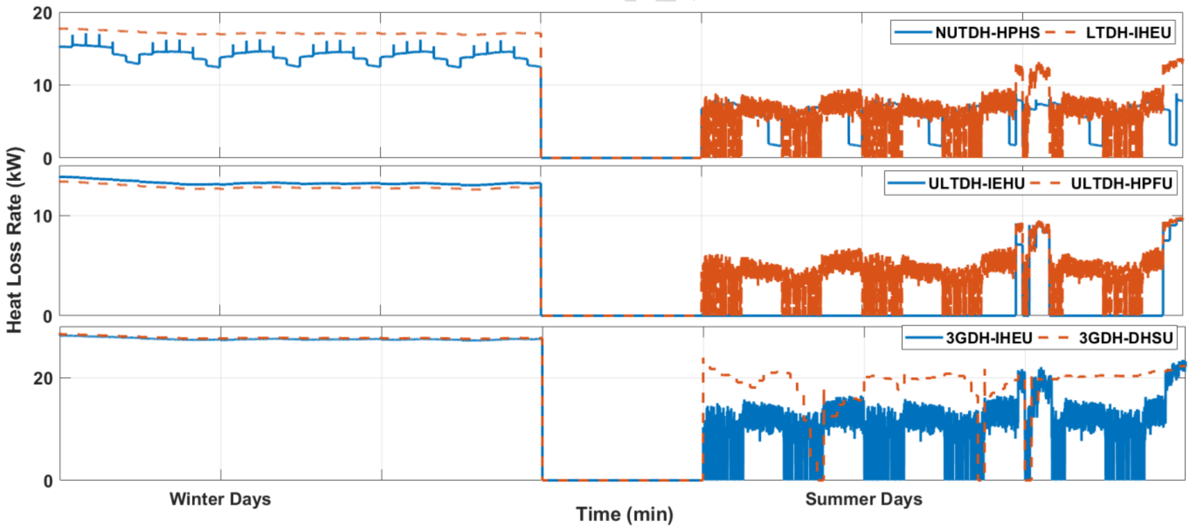

Fig. 12 The rate of heat losses in the entire pipeline of the considered scenarios for an entire year.

Expectedly, because of the low supply temperature and lower load (not covering the DHW demand), the ULTDH-IEHU system offers the lowest rate of losses among all of the systems. As such, the ULTDH-HPFU presents a very interesting rate of loss. However, these two cases are not that much of interest because the former lacks from not covering the DHW demand of the network, and the former suffers from the very high cost of the substation that makes it not be cost-effective. On the other hand, the 3GDH system in both schemes shows a very high rate of loss compared to others where the heat storage furnished scheme is even less efficient. In this between, the LTDH system offers a very good rate of loss and the NUTDH-HPHS performs even better. In fact, the NUTDH-HPHS is second best system after the ULTDH systems in terms of heat loss rate.

Table 4 gives statistics about the overall annual performance of the DH system in various scenarios. This information includes the total annual rate of heat lost from the entire pipeline in percentage and numeric formats, and the total annual work of booster pumps for overcoming the pressure losses. As the scale of piping is small in the considered case study, the last row of the table presents information about the rate of annual heat loss for various cases when being scaled to a real DH system. For this, the average values reported for realistic DH systems in the literature were used. As seen, again, expectedly, the best heat efficiency performance is for the ULTDH-IEHU with 3.5% heat loss (10.1% scaled heat loss) for preparation of the space heating demand only. Next, the ULTDH-HPFU presents 3.9% loss (11.3% scaled heat loss) for both DHW and space heating demands coverage. The third best performance is however for the proposed DH scheme of this work, i.e. the NUTDH-HPHS system which is only 4.7% (13.6% scaled heat loss rate). After the proposed system, the LTDH system is also performing quite good with a scaled heat loss rate of only (15.6%). The 3GDH schemes clearly are the two worst cases in this respect. One should note that, in fact, among all possibilities, the main competitor for the NUTDH-HPHS system is the LTDH system as the other cases are suffering from their specific disadvantages that were discussed before. Here, it is well proved that the NUTDH-HPHS system outperforms the LTDH scheme as well.

Table 4 The details of the overall performance of the various district heating cases in an entire year.

DH Scheme	3GDH		ULTDH		LTDH	NUTDH
Substation Configuration	**IHEU**	**DHSU**	**IEHU**	**HPFU**	**IHEU**	**HPHS**
Total Annual Work of Pump (MWh)	7.27	12.6	15.2	14.1	11.8	10.1
Total Annual Heat Lost (MWh)	272.85	323.3	90.8	122.4	164.6	141.5
Rate of Heat Loss for the small case (%)	8.85	10.4	3.5	3.9	5.4	4.7
The Scaled Rate of Heat Loss (%)	25.5	30.0	10.1	11.3	15.6	13.6

As such, according to the table, the NUTDH system offers a middle-range pump work among all the cases where ULTDH-IHEU demands the highest rate of work pump. Of course, the rate of required work of pump is very small compared to the values of heat losses. Thus, this factor is less of importance in the evaluations.

Figure 13 presents information about the rate of electricity consumption in each of the schemes. Among the considered cases, only ULTDH-HPFU, ULTDH-IEHU and NUTDH-HPHS systems are the only cases that depend on the electricity grids. Overall, dependency to electricity cannot be considered as an advantage. Especially, when it is in the small scale like buildings. However, when this dependency is in the form of a synergy between the sectors, it can be considered as a strong benefit because it will create a huge potential for cost-effective management of energy and for moving toward a smart energy system. According to the figure, the synergy of the NUTDH-HPHS system with the electricity network is extremely higher than the other cases as the heat pumps should periodically supply high-temperature water for both DHW and space heating uses while the other two cases only provide energy for DHW use.

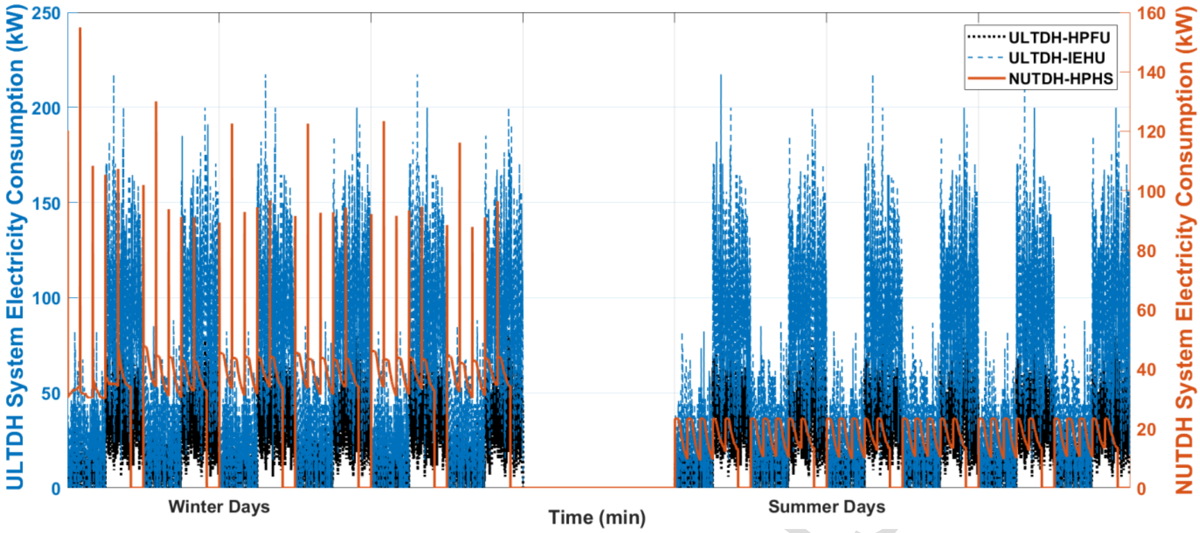

Fig. 13 Electricity consumption in each of the three electricity dependent DH schemes.

Taking into account the presented results, it can be concluded that the NUTDH-HPHS system outperforms all the competitor DH systems in an overall techno-economic point of view. A summary of the main points in this comparison assessment is:

- Both of the 3GDH schemes are extremely inefficient in terms of energy efficiency, offering high rate of heat losses.
- The ULTDH-IEHU is very efficient and offers a low rate of heat loss, however, firstly, it needs large pipes that is evidently costly and it does not cover the DHW demands.
- The ULTDH-HPFU unit offers a very good rate of loss and efficiency but it seriously suffers from the economic point of view where the individual heat pumps make the substation be far cost-inefficient [9].
- The LTDH-IHEU is probably the best choice among the five considered cases, except the NUTDH system, as it covers both DHW and space heating loads and offers a low rate of loss. However, it was proved that the NUTDH-HPHS system is even more efficient than this system as it is on an ultra-low temperature of supply during most of the day.
- Besides, the main advantage of the NUTDH-HPHS system that is the low rate of heat loss, the high potential of this system high for integration and utilization of any low-grade heat source, such as waste heat flows and low-grade renewable energy sources, and the reliable synergy that it makes with the electricity sector through the heat pumps, are another advantage of the proposed configuration.

Having said these all, some of the main features and characteristics of the NUTDH-HPHS system are discussed hereafter. An important factor in this system is the size of the storage tank required for the substations. The main criteria for sizing the storage tanks is that none of the buildings in the network should run out of hot water, with top node temperature of at least 45 ^{o}C, at any point during the year. The assessment of the results shows that a storage tank with 265 lit volume can satisfy this condition.

Figure 14 illustrates the supply temperature of three randomly selected buildings in the first, the third and the last streets of the network over the sample typical winter and summer days. This can show how the temperature drops along the path to the buildings and discloses how the supply mode changes from low- to high-temperature and vice versa. In addition, it is well seen how the ambient temperature (i.e. the season) can affect the performance of the DH system. According to the figure, the level of temperature drop is extremely higher during the summer days as the mass flow rate of the DH water through the pipes is smaller implying a lower medium velocity and larger temperature drop along the path.

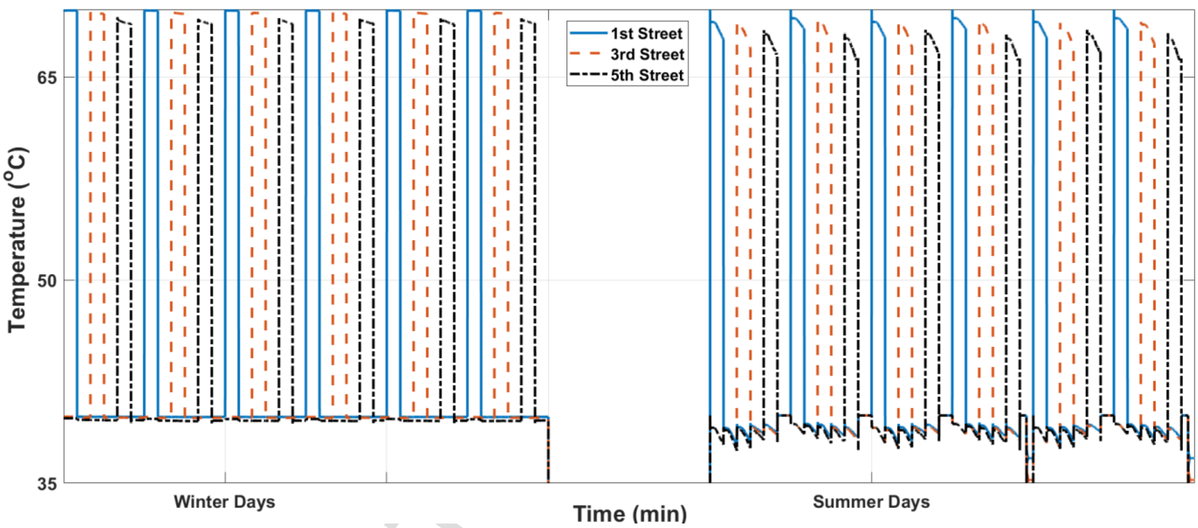

Fig. 14 Supply temperature of the buildings along the path during the typical seasonal days.

Figure 15 shows the variation of the temperature of the storage tanks (at three levels of the bottom, the middle and the top) over the sample chosen typical seasonal days for three of the buildings in the network. These buildings are located in the first, the third and the last streets. This figure shows how the storage tank is charged during the charging time and how it is discharged when the system is in low-temperature mode. As seen, the entire of the tanks approach the maximum possible level (equal to the supply level) in the charging phase and never run fully out of hot water with the least temperature of 45 ^{o}C on the top in discharging phase. This proves the proper size of the storage tanks in the system based on the considered demand profiles for the buildings in the network.

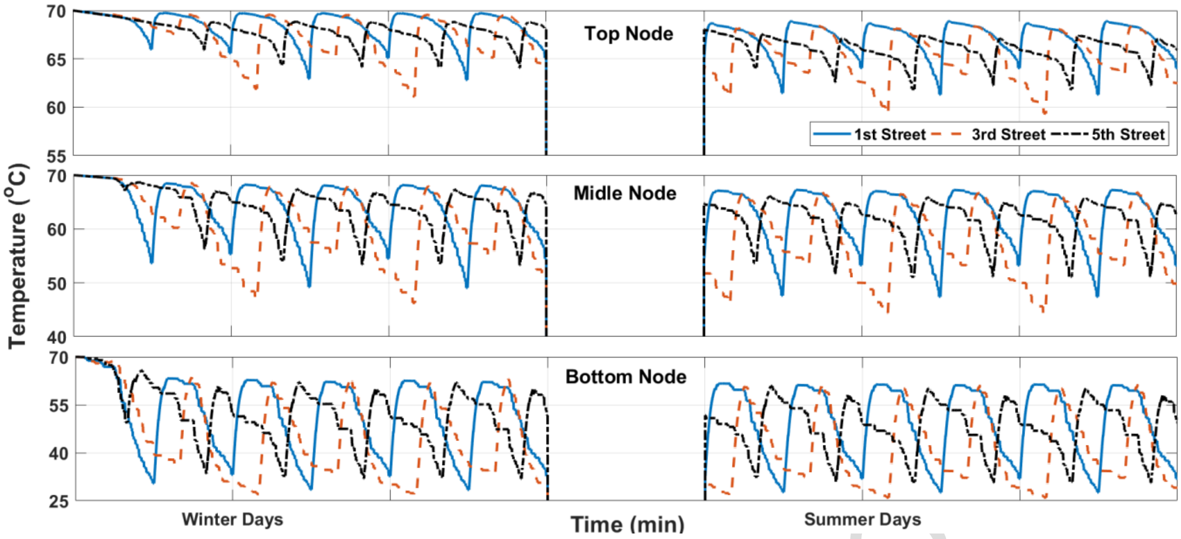

Fig. 15 Temperature of the tanks in different nodes over the sample winter and summer days.

Finally, Figure 16 shows the functioning of the control valves in the charging phase of the storage tanks. This graph presents information for three sample buildings in the first, the third and the last neighborhoods.

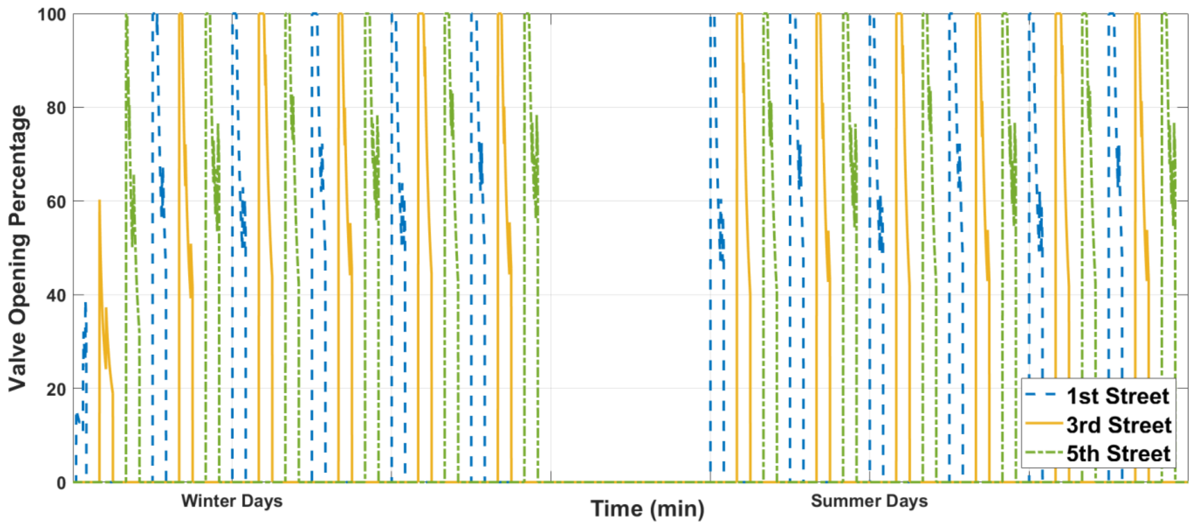

Fig. 16 The percentage of opening of the control valves in the substation of three sample buildings.

As seen, except the first typical winter day, in which the first and third control valves start with lower opening percentages (40% and 60%, respectively), the valves start with fully open condition during all the other typical summer and winter days. This shows that the storage tanks have been well discharged before starting the charging processes so that the temperature of the bottom layer of the storage tanks is quite low.

6. Conclusion

This study presents a thermodynamic analysis of an innovative concept of DH, so-called NUTDH-HPHS. This system operates based on an ultra-low temperature supply most of the time and a medium-level supply temperature during a short period of time a day. The high-temperature supply is provided through decentralized heat pumps and the DHW demand of the end-users if provided through standalone helically coiled heat storage tanks. For assessing the proficiency of the system, the system was designed, sized and thermodynamically analyzed for a small Danish case study over an entire year. Besides, the performance of the system was compared to several popular DH schemes, which were designed, sized and analyzed for the same case study. These DH technologies are LTDH-IHEU, ULTDH-IEHU, ULTDH-HPFU, 3GDH-IHEU and 3GDH-DHSU systems. The results of the analyses and comparisons prove that the proposed DH scheme is highly competent, offering a very low rate of loss and high reliability for covering the space heating and DHW demand of the network. Of the further advantages of this DH design, one could mention its high possibility for integrating low-grade renewable energy systems and the fact that it makes a concrete synergy with the electricity grid. In large-scale applications, it can be highly advantageous, bringing a huge energy-economy saving and resulting in an excellent cost-effectiveness. In addition, this DH design does not suffer from the legionella threat. As a whole, it can be concluded that the proposed NUTDH-HPHS system is a high potential DH scheme for future energy systems where the energy is supposed to be supplied through highly integrated energy systems with a high share of renewables and with excellent energy and cost efficiencies.

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- Heat pumps come into service for a district heating with non-uniform temperature.
- The system works at ultra-low temperature level for space heating most of the time.
- Medium temperature is supplied by heat pumps for domestic hot water preparation.
- The proposed system offers a low rate of heat loss and ensures no legionella risk.
- The system outperforms others in terms of an overall energy-economic performance.